Variable Displacement Compressor

BACKGROUND OF THE INVENTION

The present invention relates to a variable displacement compressor used in a vehicular air conditioner to compress refrigerant gas.

For example, a variable displacement swash plate type compressor has a cylinder block in which cylinder bores are defined. A single-headed piston is accommodated in each cylinder bore. A drive shaft is rotatably supported by a compressor housing. A lug plate is attached to the drive shaft. The lug plate rotates integrally with the drive shaft. A swash plate is supported by the drive shaft such that the swash plate slides along and inclines with respect to the drive shaft. A hinge mechanism is located between the lug plate and the swash plate.

The swash plate is coupled to the lug plate through the hinge mechanism. The swash plate integrally rotates with the drive shaft through the lug plate and the hinge mechanism. Rotation of the drive shaft is converted to reciprocation of the pistons with the lug plate, the hinge mechanism, and the swash plate. The hinge mechanism permits the swash plate to slide along and to incline with respect to the drive shaft. The displacement of the compressor is changed, accordingly. The inclination angle of the swash plate with respect to a plane perpendicular to the axis of the drive shaft decreases as a section of the swash plate about the drive shaft, or a center portion of the swash plate, approaches the cylinder block. The inclination angle increases as the swash plate center portion moves away from the cylinder block.

The compressor disclosed in Japanese Laid-Open Patent

Publication No. 5-79456 has a spring for urging a swash plate in a direction increasing the inclination angle to improve the displacement restoring performance from the minimum displacement, that is, to improve the displacement increasing performance. The compressor disclosed in Japanese Laid-Open Patent Publication No. 2001-107851 has a minimum inclination angle defining member that defines the minimum inclination angle of the swash plate by contacting the swash plate.

Specifically, a sleeve is slidably fitted about the drive shaft of the compressor disclosed in the publication No. 5-79456. The sleeve has a pivot axle. The pivot axle supports the swash plate such that the swash plate can be inclined. A snap ring is fitted about the drive shaft. The snap ring is located between the sleeve and the cylinder block. An urging spring is located between the snap ring and the sleeve. The urging spring urges the center portion of the swash plate away from the cylinder block with the sleeve in between.

In the technique disclosed in Japanese Laid-Open Patent Publication No 2001-107851, a snap ring is fitted about the drive shaft and located between the swash plate and the cylinder block. The minimum inclination angle of the swash plate is defined by contact between the center portion of the swash plate and the snap ring.

In Japanese Laid-Open Patent Publications No. 5-79456 and No. 2001-107851, a snap ring that is separate from a drive shaft is used. Accordingly, the number of components forming the compressor is increased. Further, a groove for receiving the snap ring must be machined on the drive shaft. Thus, the number of steps for machining the drive shaft is increased. This, in turn, increases the number of steps for manufacturing the compressor.

SUMMARY OF THE INVENTION

Accordingly, it is an objective of the present invention to reduce the number of components and the number of steps required for manufacturing variable displacement compressors.

To achieve the above-mentioned objective, the present invention provides a variable displacement compressor. The compressor includes a compressor housing, a cylinder block having a cylinder bore, and a single-headed piston housed in the cylinder bore. A drive shaft is rotatably supported by the compressor housing. A lug plate is attached to the drive shaft to rotate integrally with the drive shaft. A cam plate is supported by the drive shaft to slide along and incline with respect to the drive shaft. A hinge mechanism is located between the lug plate and the cam plate. Rotation of the drive shaft is converted into reciprocation of the piston with the lug plate, the hinge mechanism, and the cam plate. The hinge mechanism guides the cam plate such that the cam plate slides along the drive shaft while being inclined, and a displacement of the compressor is changed, accordingly. An urging spring urges the cam plate in a direction increasing the inclination angle of the cam plate. The drive shaft has a step formed directly on a part of the drive shaft that is between the cam plate and the cylinder block. The step includes a seat surface that intersects an axis of the drive shaft and faces the cam plate. The urging spring is located between the seat surface and the camp plate.

According to another aspect of the invention, a variable displacement compressor that includes a compressor housing and a cylinder block having a cylinder bore is provided. The compressor includes a single-headed piston housed in the cylinder bore. A drive shaft is rotatably supported by the compressor housing. A lug plate is attached to the drive

shaft to rotate integrally with the drive shaft. A cam plate is supported by the drive shaft to slide along and incline with respect to the drive shaft. A hinge mechanism is located between the lug plate and the cam plate. Rotation of the drive shaft is converted into reciprocation of the piston with the lug plate, the hinge mechanism, and the cam plate. The hinge mechanism guides the cam plate such that the cam plate slides along the drive shaft while being inclined, and a displacement of the compressor is changed, accordingly. The drive shaft has a step formed directly on a part of the drive shaft that is between the cam plate and the cylinder block. The step includes a seat surface that intersects an axis of the drive shaft and faces the cam plate. When the cam plate contacts the seat surface, an minimum inclination angle of the cam plate is defined.

Other aspects and advantages of the invention will become apparent from the following description, taken in conjunction with the accompanying drawings, illustrating by way of example the principles of the invention.

BRIEF DESCRIPTION OF THE DRAWINGS

The invention, together with objects and advantages thereof, may best be understood by reference to the following description of the presently preferred embodiments together with the accompanying drawings in which:

Fig. 1 is a cross-sectional view illustrating a piston type variable displacement compressor according to a first embodiment of the present invention;

Fig. 2(a) is an enlarged partial cross-sectional view illustrating a procedure for determining the position of an adjusting member in the compressor shown in Fig. 1;

Fig. 2(b) is an enlarged partial cross-sectional view

illustrating the procedure for determining the position of the adjusting member shown in Fig. 2(a);

Fig. 3 is a partial cross-sectional view illustrating a second embodiment of the present invention; and

Fig. 4 is a partial cross-sectional view illustrating a section including a step according to a modified embodiment of the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

First and second embodiments of the present invention will be described in the following. In the second embodiment, only the parts different from the first embodiment are explained. Like members are given the like numbers and detailed explanations are omitted.

(Variable Displacement Swash Plate Type Compressor)

Fig. 1 is a cross-sectional view illustrating a variable displacement swash plate type compressor according to the first embodiment. The left end of the compressor in Fig. 1 is defined as the front of the compressor, and the right end is defined as the rear of the compressor. The compressor includes a cylinder block 11, a front housing member 12, and a rear housing member 14. The front housing member 12 is secured to the front end of the cylinder block 11. The rear housing member 14 is secured to the rear end of the cylinder block 11 with a valve assembly 13 in between. The cylinder block 11, the front housing member 12, and the rear housing member 14 are made of an aluminum-based metal. The cylinder block 11, the front housing member 12, and the rear housing member 14 are fastened to one other with bolts 20 (only one is shown in the drawing). The cylinder block 11, the front housing member 12, and the rear housing member 14 form a compressor housing 10. That is, the cylinder block 11 forms a

part of the compressor housing 10.

The cylinder block 11 and the front housing member 12 define a crank chamber 15. A shaft member 16 is rotatably supported in the crank chamber 15. The shaft member 16 is made of an iron-based metallic material. The shaft member 16 is coupled to a vehicle drive source (external drive source), which is an engine E, through a power transmission PT. The shaft member 16 is rotated by power transmitted from the engine Eg.

The shaft member 16 is supported by the front housing member 12 with a radial bearing 18, which is a roller bearing. A shaft sealing member 19 is located between the front housing member 12 and the shaft member 16. A lug plate 21 is coupled to the shaft member 16 and is located in the crank chamber 15. The lug plate 21 rotates integrally with the shaft member 16. A thrust bearing 17 is arranged between the lug plate 21 and an inner surface 12a of the front housing member 12. A cam plate, which is a swash plate 23, is housed in the crank chamber 15. The swash plate 23 has a shaft hole 23a in the center portion. The shaft member 16 extends through the shaft hole 23a. The swash plate 23 is supported by the shaft member 16 at the inner circumferential surface of the shaft hole 23a. Thus, the swash plate 23 can slide along an axis L of the shaft member 16. Also, the inclination angle of the swash plate 23 with respect to a plane perpendicular to the axis L of the shaft member 16 can be changed.

A hinge mechanism 24 is located between the lug plate 21 and the swash plate 23. The hinge mechanism 24 couples the swash plate 23 with the lug plate 21. The hinge mechanism 24 includes support arms 36 (only one is shown in Fig. 1) projecting from a peripheral portion of the lug plate 21 and guide pins 37 projecting from the swash plate 23. A guide

hole 36a is formed in each support arm 36, and a spherical portion 37a is formed at the distal end of each guide pine 37. Each spherical portion 37a is fitted in the corresponding guide hole 36a. The hinge mechanism 24 permits the swash plate 23 to integrally rotate with the lug plate 21 and the shaft member 16. The hinge mechanism 24 also permits the swash plate 23 to incline relative to the shaft member 16 while sliding along the axis L of the shaft member 16.

The cylinder block 11 has cylinder bores 11a (only one is shown in the drawing) arranged about the axis L of the shaft member 16. The cylinder bores 11a extend through the cylinder block 11. The cylinder bores 11a extend along the axis L of the shaft member 16. A single headed piston 25 is accommodated in each cylinder bore 11a. The piston 25 reciprocates inside the cylinder bore 11a. The front and rear openings of each cylinder bore 11a are closed by the valve assembly 13 and the corresponding piston 25. A compression chamber 26 is defined inside each cylinder bore 11a. The volume of each compression chamber 26 changes as the corresponding piston 25 reciprocates. Each piston 25 is coupled to the peripheral portion of the swash plate 23 by a pair of shoes 27. The shoes 27 convert rotation of the swash plate 23, which rotates with the shaft member 16, to reciprocation of the pistons 25.

A suction chamber 28 and a discharge chamber 29 are defined in the rear housing member 14. The suction chamber 28 functions as a suction pressure zone. The suction chamber 28 is defined in a center portion of the rear housing member 14. The discharge chamber 29 is defined to surround the suction chamber 28. The valve assembly 13 has discharge ports 32 and discharge valve flaps 33, which are reed valves. Each discharge port 32 and each discharge valve flap 33 correspond to one of the cylinder bores 11a. Each discharge port 32

connects the corresponding compression chamber 26 with the discharge chamber 29, and the corresponding discharge valve flap 33 selectively opens and closes the discharge port 32. A suction valve mechanism 35, which includes a rotary valve, is located in the cylinder block 11.

In a suction stroke, each piston 25 moves from the top dead center to the bottom dead center. Accordingly, refrigerant gas in the suction chamber 28 is drawn into the corresponding compression chamber 26 through the suction valve mechanism 35. In a compression-discharge stroke, refrigerant gas that is drawn into the compression chamber 26 is compressed to a predetermined pressure as the piston 25 is moved from the bottom dead center to the top dead center. Then, the gas is discharged to the discharge chamber 29 through the corresponding discharge port 32 while flexing the discharge valve flap 33 to an open position.

(Suction Valve Mechanism 35)

The cylinder block 11 has an accommodation space 42 on the axis L of the shaft member 16. The cylindrical accommodation space 42 extends through the cylinder block 11. The cylinder block 11 has suction communicating passages 43 (only one is shown in the drawing), each of which corresponds to one of the compression chambers 26. The accommodation space 42 and each compression chamber 26 are connected to each other by the corresponding suction communicating passage 43. An end of each suction communicating passage 43 at the accommodation space 42, or an opening 43a, opens in an inner circumferential surface 42a of the accommodation space 42.

A cylindrical rotor 41 is rotatably accommodated in the accommodation space 42. The shaft member 16 and the rotor 41 are continuously arranged along the axis L of the shaft member

16. The rotor 41 is made of an aluminum-based metallic material. The rotor 41 has a front small diameter portion 41a and a rear large diameter portion 41b arranged along the axis L. The shaft member 16 has a fitting recess 16a at the rear end. The small diameter portion 41a is fitted to the fitting recess 16a. The rotor 41 is thus fixed to the shaft member 16. The shaft member 16 and the rotor 41 are integrated. The rotor 41 rotates in synchronization with rotation of the shaft member 16, or reciprocation of the pistons 25.

An outer circumferential surface 41c of the large diameter portion 41b and the inner circumferential surface 42a of the accommodation space 42, which is a wall surface of the cylinder block 11 defining the accommodation space 42, form a slide bearing. Therefore, an rear portion 16b of the shaft member 16 is rotatably supported by the cylinder block 11 with the rotor 41 in between. In this embodiment, the shaft member 16 and the rotor 41 form a drive shaft of the compressor. The shaft member 16 functions as a drive shaft main body. The large diameter portion 41b of the rotor 41 functions as a bearing of the drive shaft.

A projection 11b, which projects rearward, is formed at the rear end of the cylinder block 11. The cylindrical projection 11b extends through the valve assembly 13 into the suction chamber 28. The interior of the projection 11b is a through hole 50 for connecting the accommodation space 42 and the suction chamber 28 with each other. The through hole 50 of the projection 11b is formed cylindrical with its center being coaxial with the axis L of the shaft L. The diameter of the through hole 50 is greater than the diameter of the accommodation space 42.

The rotor 41 has a cylindrical space 44. The cylindrical space 44 is connected with the suction chamber 28 through the

through hole 50. An inlet passage 45 is formed in the sidewall of the rotor 41. The inner end of the inlet passage 45 always communicates with the cylindrical space 44. The outer end of the inlet passage 45, or an opening 45a, opens in an outer circumferential surface 41c of the large diameter portion 41b.

As the rotor 41, or the shaft member 16, rotates, the opening 45a of the inlet passage 45 intermittently communicates with the openings 43a of the suction communicating passages 43.

That is, the opening 45a of the inlet passage 45 communicates with the opening 43a of a suction communicating passage 43 that extends from one of the cylinder bores 11a that is in the suction stroke. Accordingly, refrigerant gas in the suction chamber 28 is drawn into the compression chamber 26 of the cylinder bore 11a through the through hole 50, the cylindrical space 44, the inlet passage 45, and the suction communicating passage 43. The through hole 50, the cylindrical space 44, the inlet passage 45, and the suction communicating passages 43 form a refrigerant passage between the cylinder bores 11a and the suction chamber 28. The opening 45a of the inlet passage 45 is disconnected from the opening 43a of a corresponding suction communicating passage 43 that extends from one of the cylinder bores 11a that is in the discharge stroke. In this state, the gas in the compression chamber 26 is discharged to the discharge chamber 29 through the corresponding discharge port 32 while flexing the discharge valve flap 33.

That is, by synchronously rotating with the shaft member 16, the large diameter portion 41b of the rotor 41 functions as a rotary valve that selectively opens and closes the refrigerant gas passage between the cylinder bores 11a and the

suction chamber 28.

(Structure for Limiting Sliding of Shaft Member 16 and Rotor 41)

An adjusting member 51 is press fitted in the through hole 50 of the projection 11b. The adjusting member 51 is made of an aluminum-based metallic material. A through hole 51a is formed in a center portion of the adjusting member 51 to connect the suction chamber 28 with the cylindrical space 44. A front end face 51b of the adjusting member 51 is located in the through hole 50 and faces a rear end face 41d of the rotor 41. The front end face 51b functions as a movement restriction member. That is, when the shaft member 16 and the rotor 41 are slid rearward along the axis L, the rear end face 41d of the rotor 41 abuts against the front end face 51b of the adjusting member 51. In this state, the front end face 51b prevents the shaft member 16 and the rotor 41 from being slid further rearward along the axis L.

During operation of the compressor, compression load of refrigerant gas acting on the pistons 25 act on the inner surface 12a of the front housing member 12 through the shoes 27, the swash plate 23, the hinge mechanism 24, the lug plate 21, and the thrust bearing 17. The compression load slides an integrated body including the pistons 25, the swash plate 23, the lug plate 21, the shaft member 16, and the rotor 41 frontward along the axis L. This sliding motion is limited by the inner surface 12a of the front housing member 12 with the thrust bearing 17 in between.

When frontward sliding of the shaft member 16 and the rotor 41 along the axis L is limited, a predetermined clearance X exits between the rear end face 41d of the rotor 41 and the front end face 51b of the adjusting member 51. The

clearance X corresponds to the amount of permitted sliding of the shaft member 16 and the rotor 41 along the axis L.

The clearance X is determined such that, for example, rotation of the shaft member 16 and the rotor 41 in the compressor housing 10 is permitted and displacement of contacting sections of the shaft member 16 and the shaft sealing member 19 due to sliding of the shaft member 16 and the rotor 41 is prevented. The clearance X is, for example, approximately 0.1 mm. In the drawings, the clearance X is exaggerated.

The adjusting member 51 is press fitted in the through hole 50 by a predetermined distance Y such that the clearance X is created. The distance Y represents the space between the rear end face 51c of the adjusting member 51 and an end face 11c of the projection 11b.

A procedure for determining the position of the adjusting member 51 in the through hole 50 during assembly of the compressor will now be described. First, prior to fixing the rear housing member 14 to the cylinder block 11, the adjusting member 51 is press fitted in the through hole 50 from the side corresponding to the end face 11c to determine a provisional position of the adjusting member 51 as shown in Fig. 2(a). That is, the adjusting member 51 is inserted in the through hole 50 such that the rear end face 51c of the adjusting member 51 is located frontward in a direction along the axis L from the end face 11c of the projection 11b by an amount smaller than the distance Y.

Then, using a jig 61 shown in Fig. 2(b), the position of the adjusting member 51 is determined. The jig 61 includes a flat end face 62 and a cylindrical projection 63 formed on the flat end face 62. The diameter of the projection 63 is less

than the diameter of the through hole 50 and is greater than the diameter of the through hole 51a of the adjusting member 51. The amount by which the projection 63 projects from the flat end face 62 is equal to the distance Y.

Using the projection 63 of the jig 61, the adjusting member 51 is pressed further frontward from the provisional position in the through hole 50. The jig 61 is moved relative to the projection 11b until the flat end face 62 of the jig 61 contacts the end face 11c of the projection 11b. That is, until the flat end face 62 contacts the end face 11c, the projection 63 continues pushing the adjusting member 51. Therefore, the distance Y is created between the rear end face 51c of the adjusting member 51 and the end face 11c of the projection 11b. Accordingly, the distance between the rear end face 41d of the rotor 41 and the front end face 51b of the adjusting member 51 is equal to the clearance X when frontward sliding of the shaft member 16 and the rotor 41 along the axis L is limited.

(Variable Displacement Structure of Compressor)

As shown in Fig. 1, the crank chamber 15 is connected to the discharge chamber 29 with a pressure supply passage 49. A conventional displacement control valve 52 is located in the pressure supply passage 49. The control valve 52 is an electromagnetic valve.

The shaft member 16 has a through hole 47. The rotor 41 has a hole 48. The through hole 47 is connected to the cylindrical space 44 through the rotor hole 48. A restriction 48a is formed in the rotor hole 48. The suction chamber 28 is connected to the crank chamber 15 through the through hole 51a of the adjusting member 51, the cylindrical space 44, the rotor hole 48, and the through hole 47.

Adjusting the degree of opening of the control valve 52 controls the relationship between the flow rate of high-pressure gas flowing into the crank chamber 15 through the supply passage 49 and the flow rate of gas flowing out of the crank chamber 15 to the suction chamber 28 through the through hole 47, the rotor hole 48, the cylindrical space 44, and the through hole 51a. The pressure of the crank chamber 15 is determined, accordingly. The difference between the pressure in the crank chamber 15 and the pressure in the compression chambers 26 with the pistons 25 in between is changed according to changes in the crank chamber pressure. This alters the inclination angle of the swash plate 23. As a result, the stroke of the pistons 25, that is, the discharge displacement, is controlled.

For example, as the opening degree of the control valve 52 is increased, the pressure in the crank chamber 15 is increased. As the pressure in the crank chamber 15 is increased, the swash plate 23 slides such that the center portion of the swash plate 23 approaches the cylinder block 11. That is, the inclination angle is reduced. The stroke of the pistons 25 is reduced, accordingly. This, in turn, decreases the displacement of the compressor. In contrast, as the opening degree of the control valve 52 is decreased, the pressure in the crank chamber 15 is lowered. As the pressure in the crank chamber 15 is decreased, the swash plate 23 slides such that the center portion of the swash plate 23 moves away from the cylinder block 11. That is, the inclination angle is increased. The stroke of the pistons 25 is increased, accordingly. This, in turn, increases the displacement of the compressor.

In this embodiment, the outer diameter of the rotor large diameter portion 41b is greater than the outer diameter of the

shaft member 16. Therefore, between the large diameter portion 41b of the rotor 41 and the rear portion 16b of the shaft member 16, which is adjacent to the large diameter portion 41b, a step 64 is defined. The step 64 has a seat surface 64a facing the rear end face of the swash plate 23. The seat surface 64a is a part of an end face of the large diameter portion 41b that is adjacent to the shaft member 16. That is, the step 64 is formed at the joint between the shaft member 16 and the rotor 41, which form the drive shaft. In other words, the step 64 is integrally formed with, or directly formed on, the drive shaft (the shaft member 16 and the rotor 41).

A return spring 65, which is a coil spring, is fitted about the shaft member 16. The return spring 65, which functions as an urging spring, is located between the seat surface 64a of the step 64 and the rear center portion of the swash plate 23. The return spring 65 has a fixed end located at the seat surface 64a and a free end at the swash plate 23. The seat surface 64a of the step 64 functions as a spring seat that supports the fixed end of the return spring 65.

The height of the step 64 in the radial direction of the shaft member 16, or the difference between the radius of the outer circumference of the rear portion 16b of the shaft member 16 and the radius of the outer circumferential surface 41c of the large diameter portion 41b is greater than the diameter of the steel wire of the return spring 65, or the diameter of the wire forming the spring 65.

The rear center portion of the swash plate 23 contacts the free end of the return spring 65. The center portion of the swash plate 23 is urged in a direction away from the cylinder block 11, or in a direction increasing the inclination angle, by the force of the return spring 65. When

the swash plate 23 is moved to the minimum inclination position by fully opening the control valve 52, the pressure in the crank chamber 15 is lowered by decreasing the opening degree of the control valve 52. In this state, the force of the return spring 65 permits the inclination angle of the swash plate 23 to be quickly increased.

The above embodiment provides the following advantages.

(1) The step 64 is directly formed on the drive shaft, which is formed by assembling the shaft member 16 and the rotor 41, and the seat surface 64a of the step 64 functions as the spring seat of the return spring 65. Therefore, the present invention requires no snap ring such as the one disclosed in Japanese Laid-Open Patent Publication No. 5-79456. Since no snap ring is required, the number of components and the number of steps for manufacturing the compressor are reduced.

(2) The rotor 41 has the outer circumferential surface 41c, which functions as a part of a slide bearing, and forms a part of the drive shaft. That is, the bearing for rotatably supporting the drive shaft, or the shaft member 16 and the rotor 41, is a simple slide bearing. Therefore, compared to a case where roller bearing is employed, the structure of the compressor is simplified.

(3) The step 64 is formed by the difference between the diameter of the rear portion 16b of the shaft member 16 and the diameter of the large diameter portion 41b of the rotor 41, which is greater than the diameter of the rear portion 16b. The greater the outer diameter of the large diameter portion 41b is, the lower the surface pressure of the outer circumferential surface 41c becomes. Also, a greater outer diameter of the large diameter portion 41b increases the speed

of the outer circumferential surface 41c, or the circumferential velocity. Generally, in a slide bearing, an oil film is likely to be squeezed out at a high load and low speed rotation. Therefore, even if the compressor is operating at a low speed under a high load, the load on the outer circumferential surface 41c of the rotor 41 is relatively low and the speed of the outer circumferential surface 41c is relatively high. This prevents the oil film between the outer circumferential surface 41c and the inner circumferential surface 42a of the accommodation space 42 from being squeezed out.

That is, by increasing the diameter of the large diameter portion 41b and the diameter of the accommodation space 42, the durability of the slide bearing, which is formed by the outer circumferential surface 41c and the inner circumferential surface 42a, is increased. Further, one of the by-products of the improved durability is that the step 64 is formed by the difference between the diameter of the large diameter portion 41b and the diameter of the rear portion 16b. That is, the diameter difference is efficiently used for forming the step 64.

(4) The shaft member 16 and the rotor 41, which form the drive shaft, are separate from each other. Therefore, for example, the shape, the measurement, and the material of the rotor 41 are not limited by the machining and functional conditions of the shaft member 16. Therefore, the shape, the measurement, and the material of the rotor 41 can be determined giving a higher priority to the performance of the slide bearing.

(5) The rotor 41 functions not only as a bearing but also as a rotary valve. The integration of the bearing function and the rotary valve function in the rotor 41 reduces the

number of the components of the piston type compressor.

(6) The height of the step 64 is greater than the diameter of the steel wire of the return spring 65. Therefore, the seat surface 64a stably supports the return spring 65.

A second embodiment will now be described with reference to Fig. 3. This embodiment is different from the first embodiment in that the return spring 65 is omitted. The large diameter portion 41b of the rotor 41 is extended toward the swash plate 23. That is, the step 64 is located frontward than in the first embodiment. The minimum inclination angle of the swash plate 23 is defined by contact between the swash plate 23 and the seat surface 64a of the step 64. That is, the seat surface 64a of the step 64 functions as a minimum inclination angle defining member.

This embodiment provides the same advantages as the advantages of the first embodiment.

The invention may be embodied in the following forms.

In the illustrated embodiments, the large diameter portion 41b of the rotor 41 has a diameter greater than that of the rear portion 16b of the shaft member 16. That is, the step 64 is formed between the large diameter portion 41b and the rear portion 16b. However, as shown in Fig. 4, the outer diameter of the large diameter portion 41b may be equal to the diameter of the rear portion 16b of the shaft member 16. In this case, the step 64 is formed by a brim. The brim is directly formed on the large diameter portion 41b. The brim is located on a part of the outer circumferential surface 41c that is exposed in the crank chamber 15.

In illustrated embodiments, the swash plate 23 is directly supported by the shaft member 16, which extends through the shaft hole 23a. However, a sleeve may be located between the swash plate 23 and the shaft member 16. The sleeve is loosely fitted about the shaft member 16 to slide along the shaft member 16. The sleeve has a pivot axle. The swash plate 23 is supported by the pivot axle such that the swash plate 23 can be inclined.

Therefore, when a sleeve is added to the compressor of the first embodiment, the return spring 65 is located between the sleeve and the seat surface 64a. That is, the free end of the return spring 65 contacts the sleeve, not the swash plate 23. The return spring 65 urges the sleeve away from the cylinder block 11. As a result, the swash plate 23 is urged in a direction increasing the inclination angle. When a sleeve is added to the compressor of the second embodiment, the swash plate 23 does not contact the seat surface 64a of the step 64. Instead, the sleeve contacts the seat surface 64a to define the minimum inclination angle of the swash plate 23.

In the illustrated embodiments, the compressor employs the suction valve mechanism 35, which is a rotary valve. However, the present invention may be applied to a compressor having reed valve type suction valve mechanism provided in the valve assembly 13. That is, the rotor 41 may be deprived of the rotary valve function and may function only as a bearing portion.

In the illustrated embodiments, the shaft member 16 and the rotor 41 are separate components. However, the shaft member 16 and the rotor 41 may be an integrated component.

The present examples and embodiments are to be considered

as illustrative and not restrictive and the invention is not to be limited to the details given herein, but may be modified within the scope and equivalence of the appended claims.